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THE SES LIFT FAN EVALUATION RIG

by

John M. Durkin

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AVIATION AND SURFACE EFFECTS DEPARTMENT

ASED 347

February 1977

DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

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A test rig for the experimental evaluation of Surface Effect Ship (SES) lift fan models has been designed, fabricated, and integrated into one of the 8- by 10-foot subsonic wind tunnels at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC). Centrifugal and axial flow lift fan characteristics have been investigated under steady state as well as time-varying flow rate conditions. Fans of about 2 feet

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NOTATION

Symbol	Definition	
A	Area	ft^2
C_P	Specific heat	Btu/lbM°R
d_T	Impeller tip diameter	in.
H	Head rise	ft-lb/lbM or ft
h	Enthalpy	Btu
k	Loss coefficient	
L	Torque	ft-lb
N	Shaft rotational speed	rpm
P	Pressure	lb/ft^2
Q	Volume flow rate	ft^3/sec
R	Gas property	ft-lb/lbM°R or ft/°R
R_f	Ram recovery coefficient	
SHP	Shaft power	hp
T	Temperature	°R
U_T	Impeller tip speed	ft/sec
V	Velocity	ft/sec
$\dot{W}$	Weight flow rate	lbM/sec
β	Humidity	lb water vapor/lb air
γ	Ratio of specific heats for air	
η	Efficiency	
ρ	Mass density	$lbM\text{-}sec^2/ft^4$

τ	Torque coefficient
ϕ	Flow coefficient
ψ	Head coefficient

Subscripts

n	Station number
s	Static
T	Total

ABSTRACT

A test rig for the experimental evaluation of Surface Effect Ship (SES) lift fan models has been designed, fabricated, and integrated into one of the 8- by 10-foot subsonic wind tunnels at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC). Centrifugal and axial flow lift fan characteristics have been investigated under steady state as well as time-varying flow rate conditions. Fans of about 2 feet in diameter (or less) and up to 150 hp can be handled on the rig. Instrumentation includes transducers to measure pressure, power, and efficiency. Associated with the rig is a digital data acquisition system capable of sampling data at rates of up to 12 KHZ.

ADMINISTRATIVE INFORMATION

The work presented herein was performed for the Naval Sea Systems Command, Surface Effect Ships Project Office (PMS 304) under Program Element 63534N, Task Area S4629, and DTNSRDC Work Unit 1-1630-018.

This work was conducted prior to adoption of a metric unit policy. In the interest of time and economy, no conversion to metric units has been made.

INTRODUCTION

Lift fans refer to the particular machinery on a surface effect ship which pumps relatively large quantities of air from the atmosphere to the cushion plenum and seals. In general, the fans operate at low to moderate speeds, compared with the speed of sound of air at sea level;

and, while the increase in air density cannot be ignored, it is not as great as is encountered in other types of turbomachinery such as compressors or turbochargers. The force of the pressurized air in the plenum, acting on the hull, provides aerostatic support for the ship. Depending on the particular ship design, the mean cushion pressures vary from 50 to over 400 pounds per square foot. For conventional service, fans are designed to meet a definite pressure rise and volume flow requirement and to operate at steady state conditions. On SES, however, the fans must be designed to meet a flexible design point since the required cushion pressure is reduced as the craft burns fuel during its mission. Furthermore, the instantaneous cushion pressure and volume flow rate are continually varying due to the interactions between the ship and the seaway.

It is standard practice to test fans for steady state operation, and various codes define these test procedures in precise detail.[1] These investigations yield the steady state performance maps of the static and total pressure rise, static and total efficiency, and the horsepower plotted against the corresponding volume flow rate. Figure 1 is an example of the results of these tests. Often, the fans which are tested are scale models of larger fans, or prototypes of a family of fans of various sizes. In addition, it is common to test a fan at only one speed which may differ

[1] "Test Codes for Air Moving Devices," AMCA Standard 210-67, (Air Moving and Conditioning Association, Arlington Heights, Illinois (1967).

from the design speed for a given duty. In cases such as these, the fan performance data from the steady state test can be presented in normalized (quasi-dimensionless) form or in dimensionless form. The performance curves of Figure 1 describe the performance of a particular fan, characterized by its geometry and its size, operated at a particular speed. In Figure 2, the normalized data defines the performance of the test fan at any speed, since the individual speed-dependent, dimensional performance curves collapse onto one curve when the date are divided by the appropriate power of the shaft speed. In Figure 3, the performance data of Figure 1 has been reduced to dimensionless values using basic similarity relationships. The fan pressure rise is expressed in terms of the change in the adiabatic head, and from the latter term, the head coefficient is calculated. The flow rate is similarly reduced to the dimensionless flow coefficient, as is the fan horsepower. These dimensionless curves define the performance of all fans that are geometrically similar to the test fan regardless of their size.

In dynamic tests of fans, the flow rate through the system is rapidly altered with respect to time; correspondingly, the pressure rise through the fan varies with time. In the particular application of fans to SES, the fans are very definitely operating in a dynamic environment. In addition, it is possible that, with a momentary shutoff of the leakage through the system coupled with a rapid rise in cushion pressure due to compression of the plenum volume, the fan actually experiences reverse flow; that is,

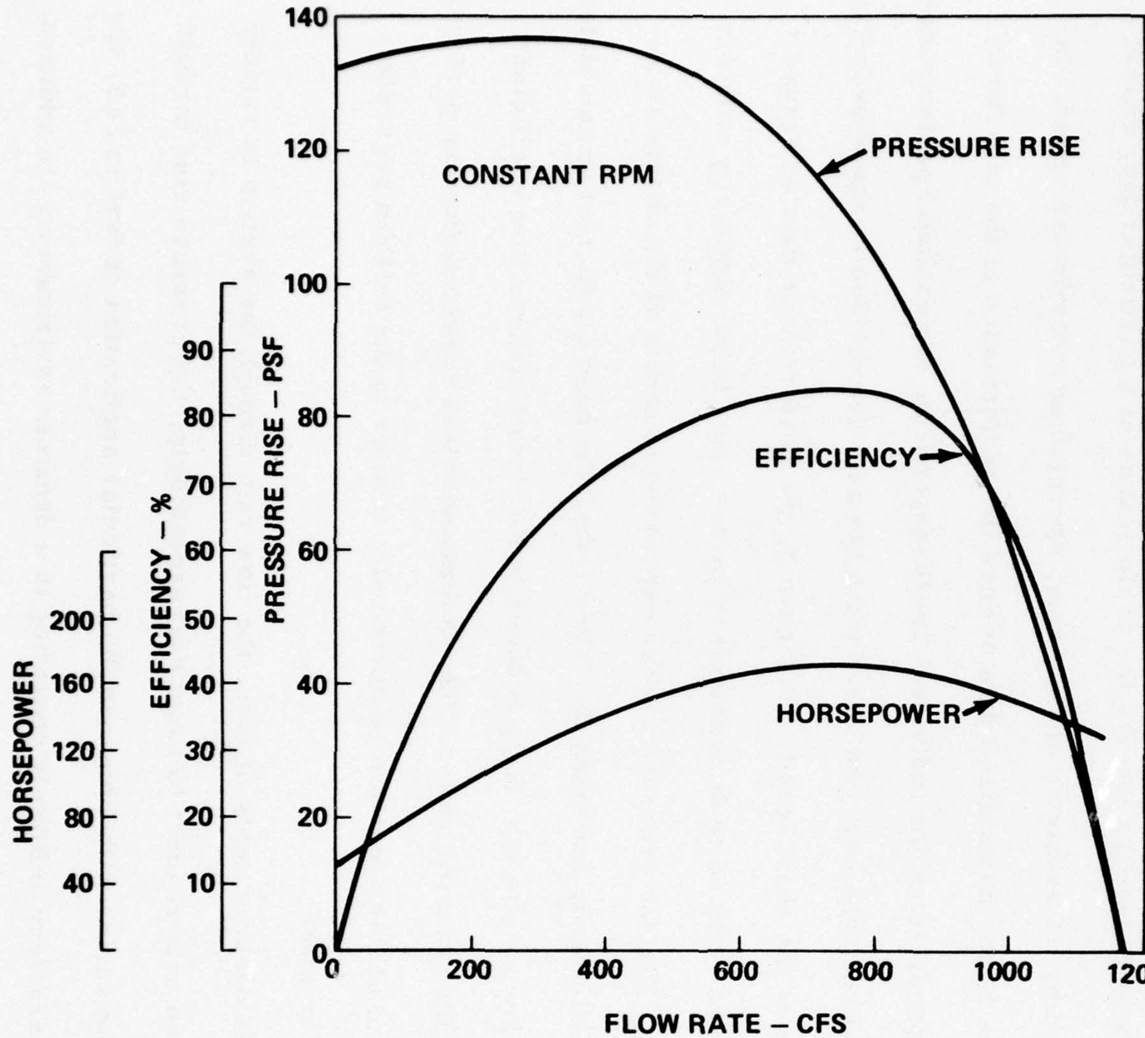

Figure 1 – Dimensional Steady-State Fan Characteristics

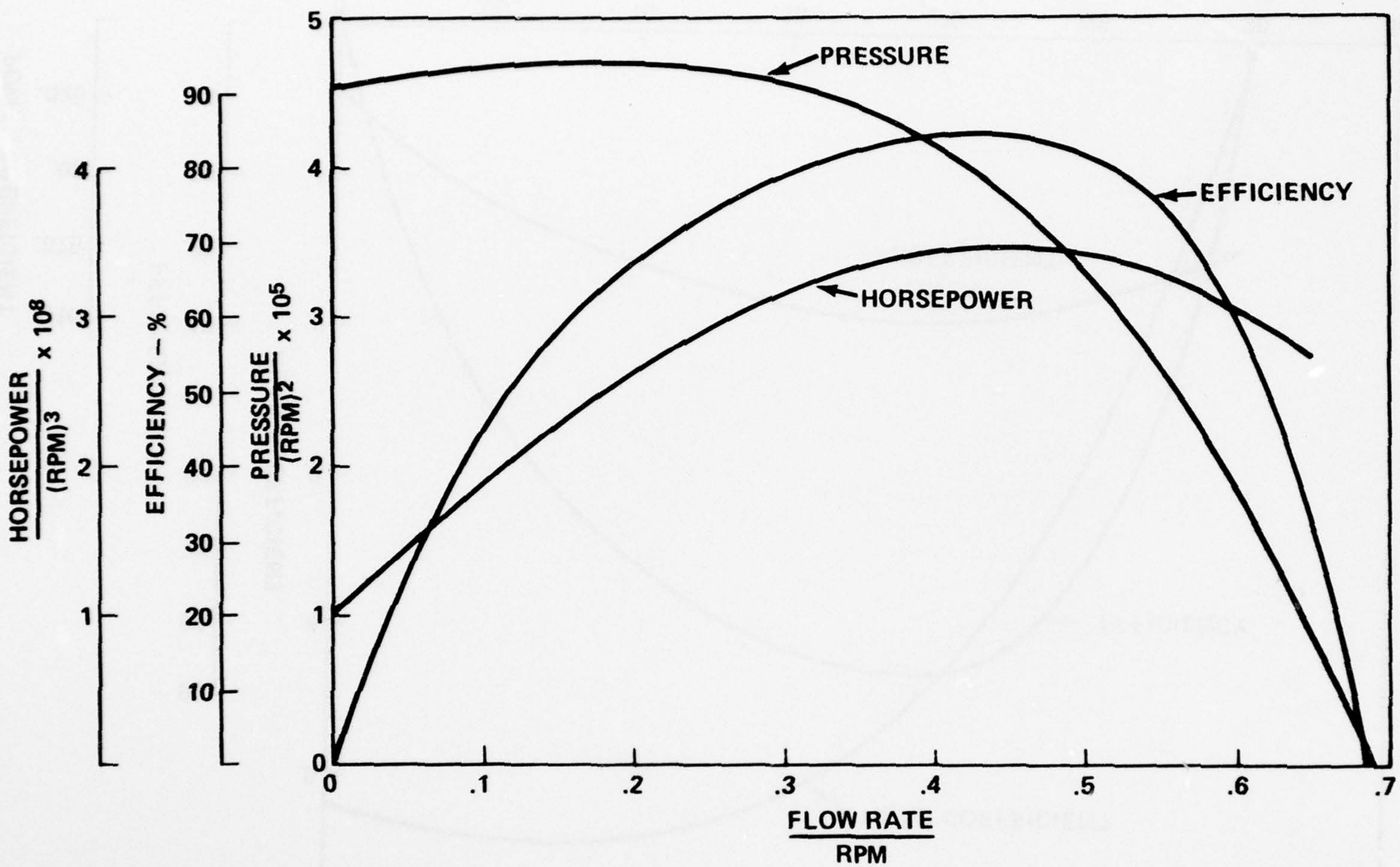

Figure 2 – Normalized Steady-State Fan Characteristics

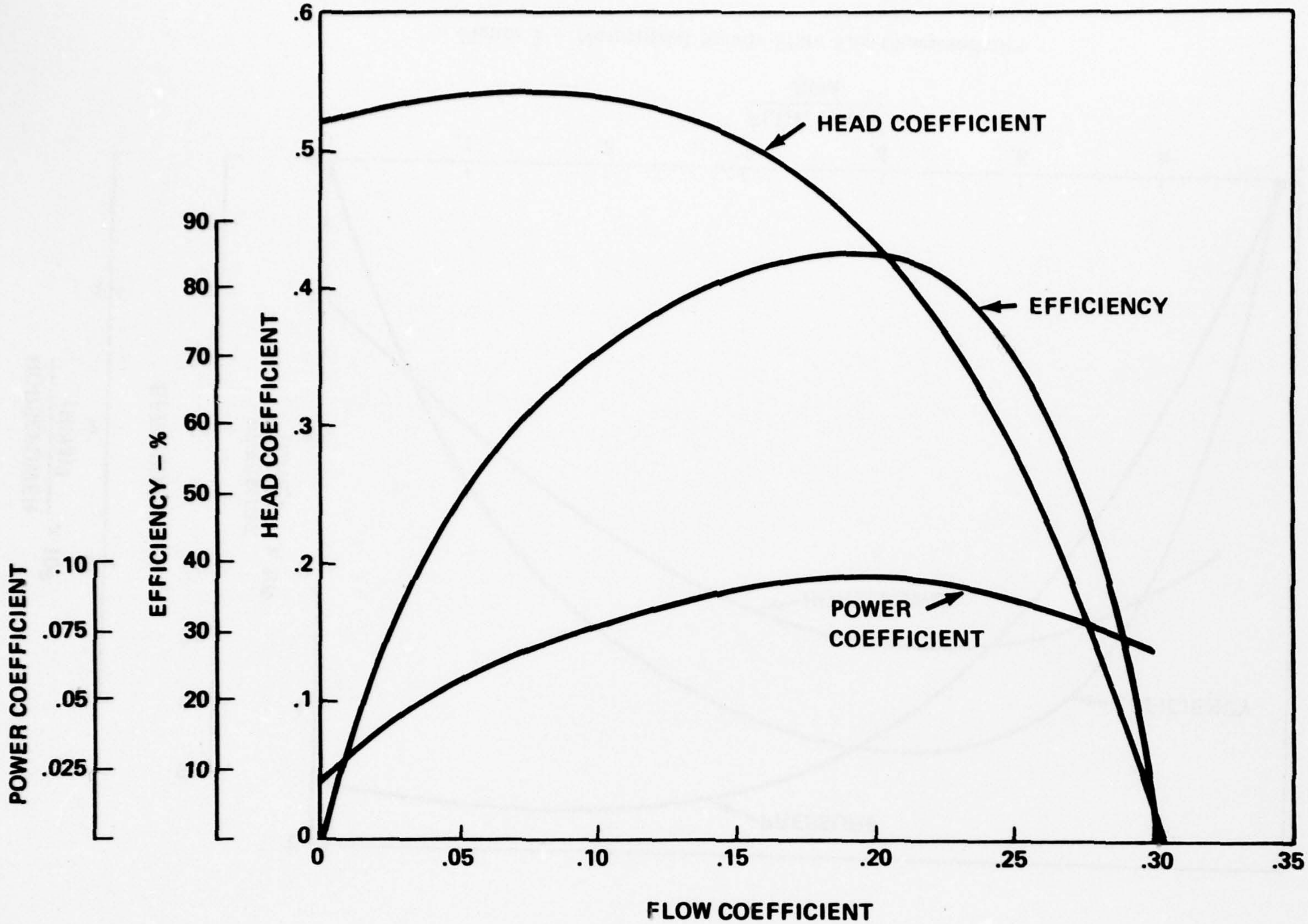

Figure 3 – Dimensionless Steady-State Fan Characteristics

air flows from the cushion through the fan to the atmosphere -- a condition referred to as "backflow."

The need to simulate the dynamic nature of the fan flow of an SES system in a controlled environment ultimately led to the construction of a fan evaluation rig at DTNSRDC. This rig consists basically of a large plenum which is pressurized by the flow from the test fan. A large exhaust duct channels the flow from the plenum back to the atmosphere. A smaller auxiliary duct is used to introduce high pressure air into the plenum in order to force backflow through the fan. The fan inlet draws air from a wind tunnel test section; thus it is possible to investigate ram effects, including ram recovery by simultaneously operating the fan and the wind tunnel. Figure 4 is an artist's concept drawing of the DTNSRDC fan evaluation rig.

DESCRIPTION OF THE FAN EVALUATION RIG

PLENUM

The DTNSRDC fan evaluation rig plenum is a box (8 feet 10 inches long by 12 feet 10 inches wide) with height adjustable between 5 1/2 and 7 feet) yielding between 580 and 820 cubic feet overall volume. The box is constructed of plywood backed by a welded aluminum frame (Figure 5). It is designed to withstand differences of 500 pounds per square foot.

As previously stated, the height of the plenum is adjustable; the ceiling fits within the vertical side walls of the plenum. An inflatable

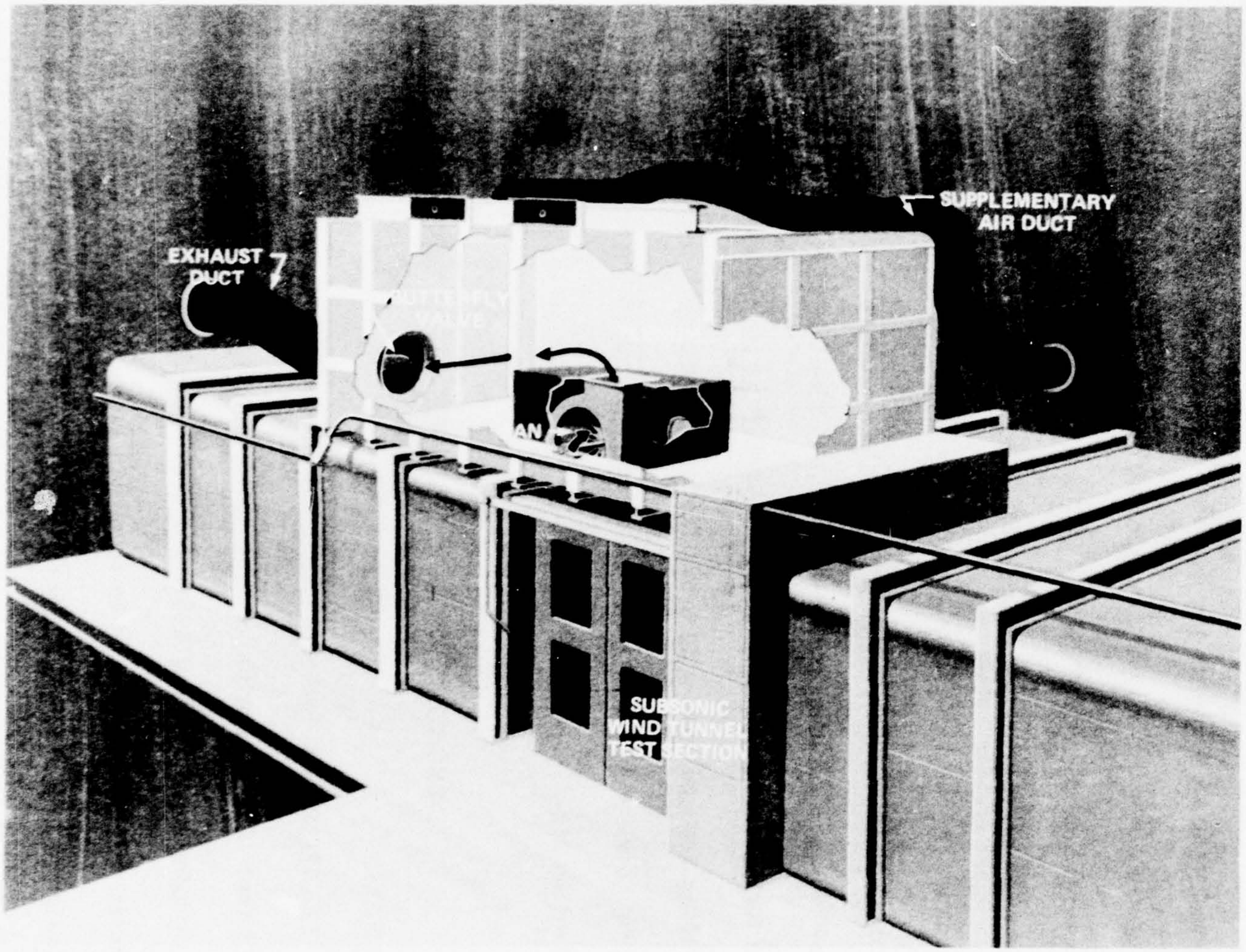

Figure 4 – Artist's Sketch of the Fan Evaluation Rig

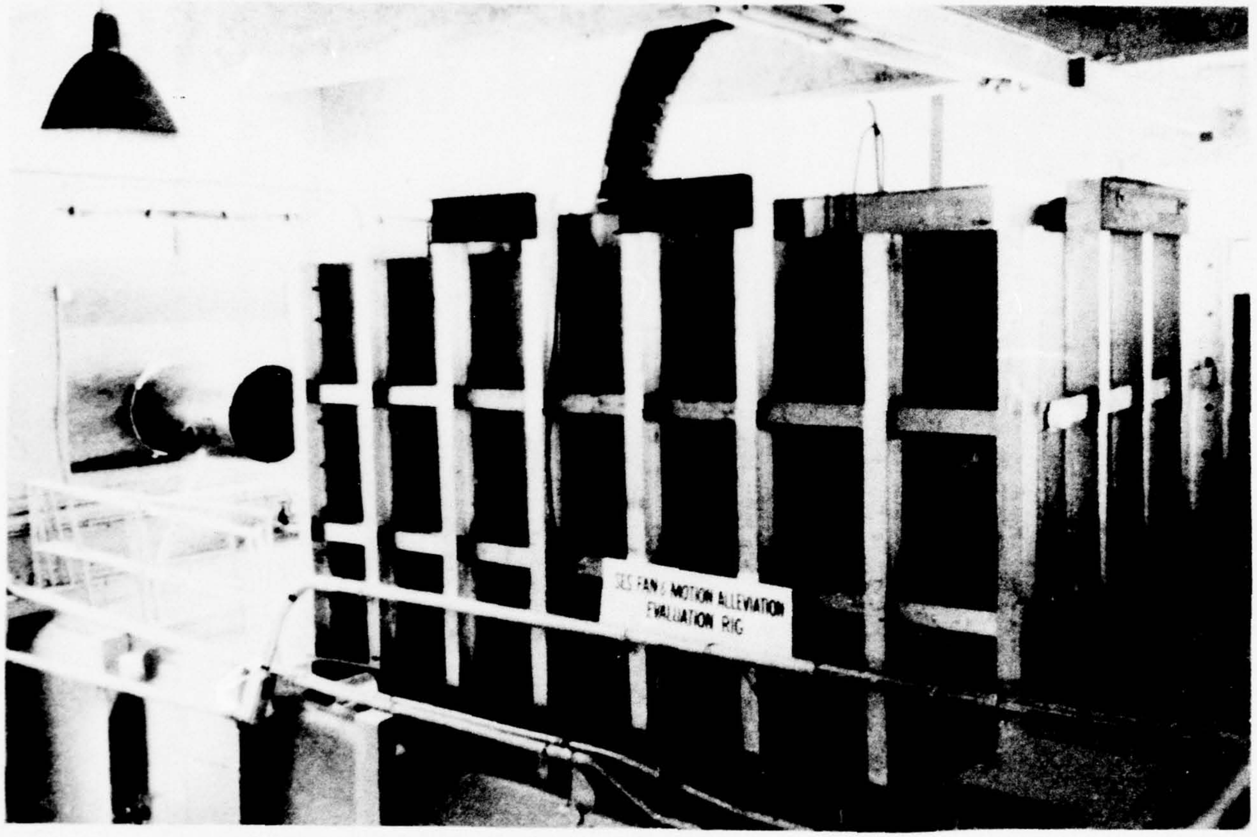

Figure 5 – Exterior Wall Detail of the Fan Evaluation Rig

tube at the corner between the ceiling and the walls aids in preventing leakage from the plenum. The test fan is mounted in, or adjacent to, a 40- by 60-inch opening in the floor of the plenum. This opening is into a subsonic wind tunnel test section, and air is drawn from the test section to the fan. This will be discussed in greater detail subsequently.

EXHAUST DUCT

The pressurized air in the plenum is ducted from the plenum to the atmosphere through a 30-inch diameter circular duct. The flow into the exhaust duct is modulated using an array of louver vanes similar to a venetian blind window shade. The exhaust valve position is controlled by an electrohydraulic servovalve operated by the test technician from the control room. Within the exhaust duct there is a station for measuring the flow using a sharp edge orifice plate. A honeycomb flow straightener is installed several pipe diameters upstream of the orifice plate to insure that a uniform flow pattern exists at the orifice. Two exhaust fans located at the downstream end of the duct are used to reduce the system resistance in order to investigate the high flow regime of the fan characteristic.

SUPPLEMENTARY SUPPLY DUCT

A 12-inch diameter duct connects the plenum to a jet compressor-type air ejector. This device is used to deliver a quantity of air at pressures to 3.5 psig. An orifice station is included in this duct. Using the air

ejector, with the primary exhaust valve closed, it is possible to raise the pressure in the plenum to values in excess of the shutoff (zero flow) pressure rise of the fan and thus force backflow through the fan. In addition, a butterfly valve is used to modulate the flow in this duct. Provision has been made to disconnect the supplementary duct from the air ejector and pipe the air from the test plenum to the atmosphere. Use of the 12-inch duct in this manner provides a more accurate control and measurement of the fan flow at flow rates less than 100 cubic feet per second than does the primary exhaust duct.

FLOW CONTROL VALVES

The flow into the primary exhaust duct is controlled using an array of louvers, as has been previously mentioned. The louvers are shown in Figure 6. The individual vanes feature a double wedge-shaped cross-section, with a rubber strip running the length of the vane, clamped at each edge of the vane. The individual vanes are connected to a control actuating rod. Pins connecting the individual louvers to the actuating rod can be removed and the louvers fixed in position. A hydraulic actuator provides the force required to move the central rod and the louvers. The actuator motion is controlled using a Moog servovalve which causes an actuator displacement that is proportional to an externally applied voltage. The assembly includes displacement transducers which indicate the louver angle, as well as providing the feedback signal for the servovalve system.

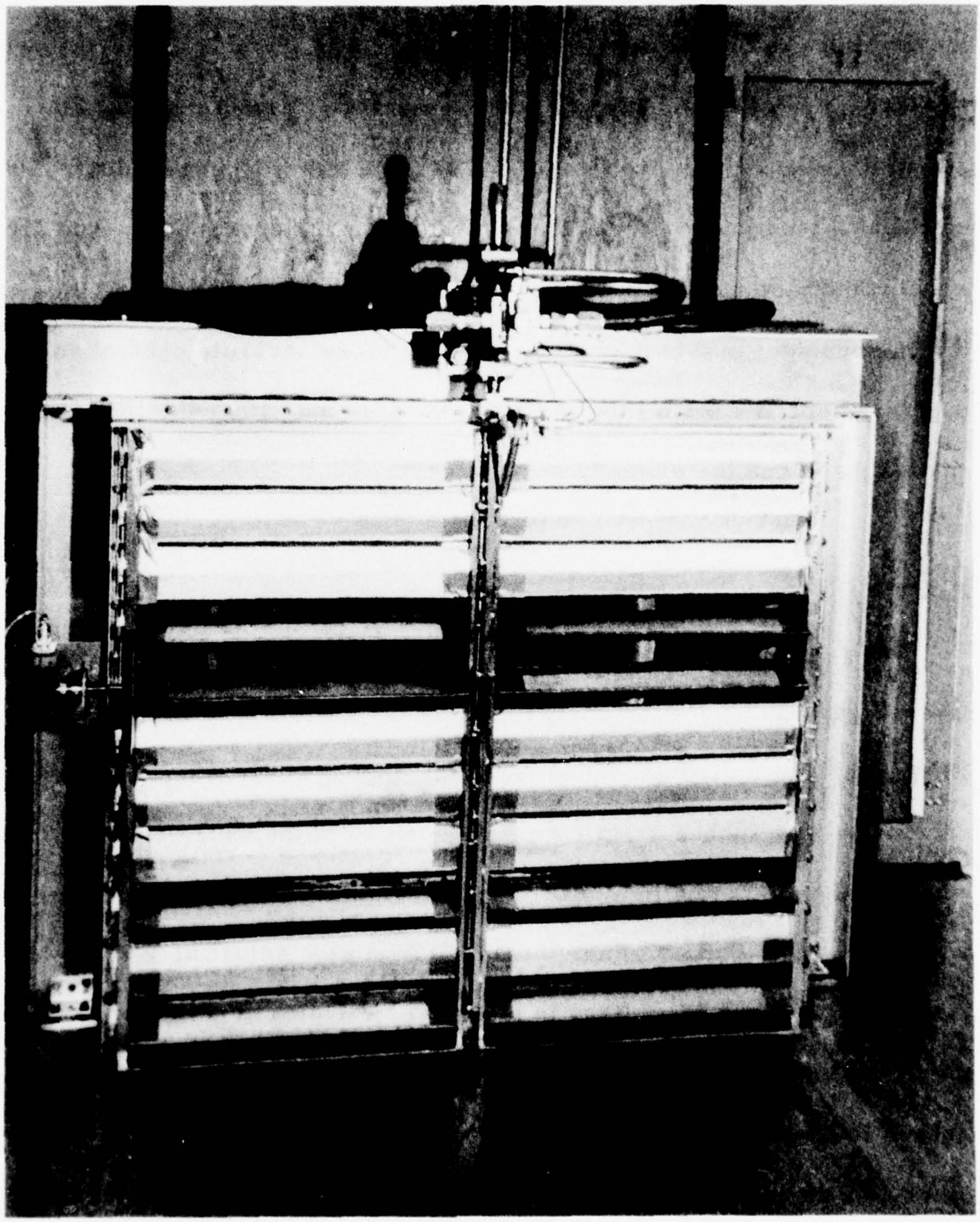

Figure 6 – Louver-Type Flow Control Valve

WIND TUNNEL

The fan evaluation rig is mounted on top of Subsonic Wind Tunnel 2.[2] The wind tunnel is capable of speeds from 10 to 140 knots and has an atmospheric test section; at maximum speed, the ram pressure is 63 pounds per square foot. Between the wind tunnel and the fan rig is an interface opening measuring 40 by 60 inches. The fan model is usually located in, or adjacent to, this opening. Naturally, the fan inlet must be sealed from the plenum; otherwise, the pressurized air is recirculated back to the inlet of the fan, with a corresponding adverse effect on the performance measurement.

MODEL POWERING

Fan models are powered by variable frequency model motors. The Aviation and Surface Effects Department has motors rated at 75 or 150 horsepower at speeds of 8,000 rpm. The electrical power to the motors is provided by an 816 KVA motor generator set which operates from 0 to 400 Hz. For cases where more than 150 hp are required, a second 150 hp motor can be operated in parallel with the first, to a total power level of about 225 hp.

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[2]Kidd, M. A., "Subsonic Wind Tunnel Facilities," NSRDC Report 3782 (Jan 1972).

INSTRUMENTATION

The instrumentation system utilized measures the torque, shaft speed, pressures, strains, valve positions, and accelerations. The electrical signals are converted to engineering units using the appropriate engineering unit per volt calibration factor determined prior to testing. The signals are digitized and recorded on digital magnetic tape; additionally, selected signals are recorded on analog tape for subsequent spectral or other analysis and/or recorded on a direct write oscillograph. The supplementary supply duct features a butterfly valve used to modulate the flow. Normally, the position of this valve is fixed by the test operating personnel prior to a test run. During dynamic tests, the valve is driven by a D.C. motor and can be operated continuously at frequencies up to 6 Hz.

Fan Shaft

Shaft torque supplied to the fan and shaft speed are both measured using Lebow shaft torque sensors which contain a 4-arm strain gage device and a 60 pulse per revolution magnetic pickup, which is an integral part of the Lebow sensor. A frequency-to-voltage converter converts the electrical signal from the shaft speed sensor to a D.C. voltage for the data acquisition system.

Pressure

Pressure measurements are made using variable reluctance differential pressure transducers. The transducer is hooked up to static pressure taps

located in the wall of circular measurement stations on the fan, viz. at the fan inlet, or to kiel-type total pressure probes located in the air stream at the measurement stations. Gage capacities are tailored to the specific test, and standard diaphragms are used.

Temperature

Iron constantan thermocouples are used for temperature measurement. These are connected to 125 F junction bridges.

Valve Position

The position of the exhaust and supplementary air supply duct valves is recorded using continuous rotating potentiometers connected directly to the respective valve shaft.

Impeller Strains

Impeller strains, for instance, blade strains, are measured using surface-mounted strain gages with one active arm. Blade strain measurements during fan testing are generally supplemented with an impeller model survey to identify impeller natural frequencies. By performing a spectral analysis of the strain data, it is possible to identify any modes which may be excited in the fan, especially during the dynamic tests.

The above represent the primary measurands for fan tests. In addition to these, system vibrations are monitored for safety reasons, usually by placing accelerometers on the shaft bearing housings and/or

on the fan housing. Finally, ambient barometric pressure, ambient temperature, and humidity in pounds of water vapor per pound of dry air are recorded manually for input to the data acquisition system.

DATA ACQUISITION

The primary data acquisition device utilizes a Raytheon 704 digital data acquisition system and processing unit, shown in Figure 7. This system samples data at a channel rate of up to 100 samples per second for 120 channels and stores these on digital magnetic tape, in addition to providing engineering unit printout for on-line analysis. The magnetic tape is used for further computer processing where plotting or comparison with other data is desired. The 100 samples per second rate is more than sufficient for sampling the varying parameters during dynamic testing.

In addition to the digital data acquisition system, an FM magnetic tape recorder and a direct write oscillograph are used. The tape recorder is for intermediate storage of high response data which can be played back to other recording instruments for spectral or time history analysis. The oscillograph provides a direct, continuous, visual readout during testing.

DATA REDUCTION AND PROCESSING

The following assumptions are used for data normalization:

- The mixture of water vapor and air behaves as a perfect gas.
- The flow processes are irreversible and adiabatic.

- The duct loss coefficients are assumed constant and independent of Mach number or Reynolds number.
- The compressibility between the fan inlet and exhaust is included.

The basic equations used in fan data reduction are the isentropic flow relationships, the equation of state, and the energy equation for a perfect gas. The fan performance is normalized to the impeller tip velocity. Derivation of the data reduction equations is shown in the appendix.

Final data processing can be performed on the CDC 6400 computer. Both tabulated output and plots of the fan performance parameters are generated during this processing.

FLOW CONTROL VALVE ADMITTANCE CHARACTERISTICS

The effective area of the primary exhaust valve from the plenum is defined as the product of the actual area and the orifice coefficient. This has been evaluated using the following procedure. The pressure difference ΔP is the difference between the plenum pressure P_p and the duct pressure P_d, and the density ρ is the weight density of the pressurized plenum air. The weight flow rate of the air in the duct is determined from measurements made at the orifice and is assumed to be constant at any station in the duct. From this, the effective area is the weight flow rate divided by the square root of the product of two times the pressure difference ΔP, the density,

and the acceleration due to gravity (32.174 feet per second per second). The equation expressing this is:

$$A = \frac{\dot{W}}{(2\rho g \Delta P)^{\frac{1}{2}}}$$

Results of this evaluation are presented in Figure 8, which shows the effective flow area versus the louver angle for two and four vanes being opened and all others fixed in the closed position.

LIFT SYSTEM OPERATION

An SES lift system includes, in addition to the lift fan, the air inlet and associated ducting to deliver ambient air to the fan inlet and the ducting from the fan to the cushion and seals. Figure 9 illustrates such a system schematically. The system pressure rise can be expressed very simply as

$$P_{s_4} - P_{s_o} = \tfrac{1}{2} \rho V_o^2 - P_{L_{0,2}} + \Delta P_{T_{2,3}} - P_{L_{3,4}} \qquad (1)$$

that is, as the sum of the ram head and the pressure rise across the fan minus inlet and diffuser losses. The equation can be expressed in terms of the system head rise as

$$H_{s_4} - H_{s_0} = \frac{V_o^2}{2g} - H_{L_{0,2}} + \Delta H_{T_{2,3}} - H_{L_{3,4}} \qquad (2)$$

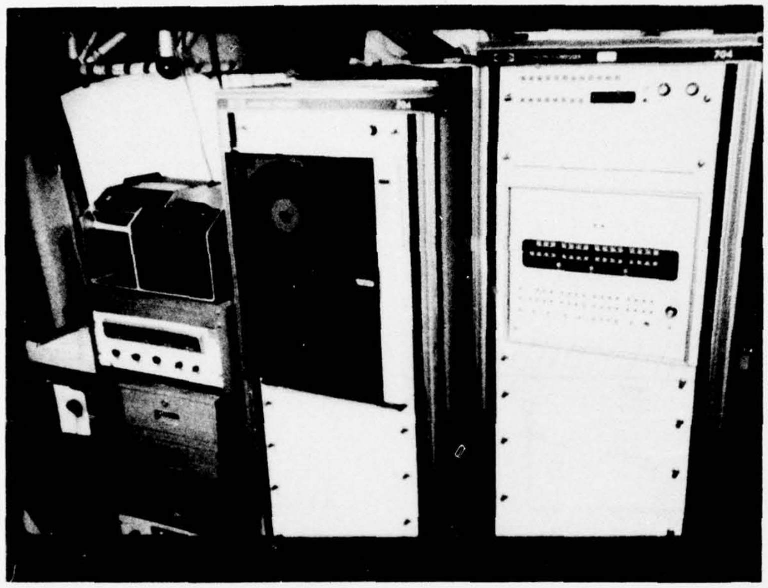

Figure 7 – Raytheon 704 Digital Data Acquisition System

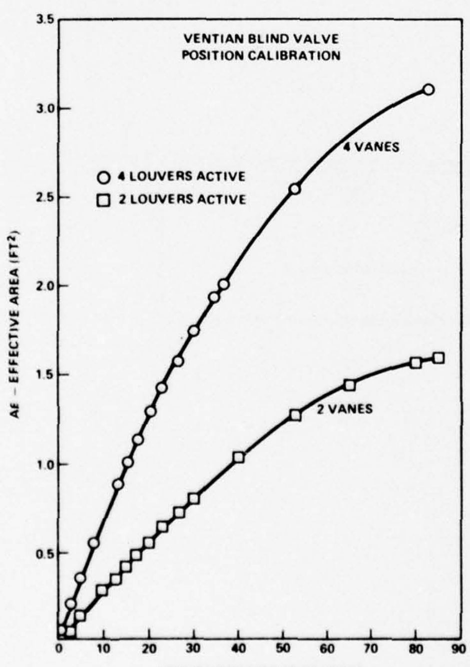

Figure 8 – Flow Control Valve Admittance Characteristics

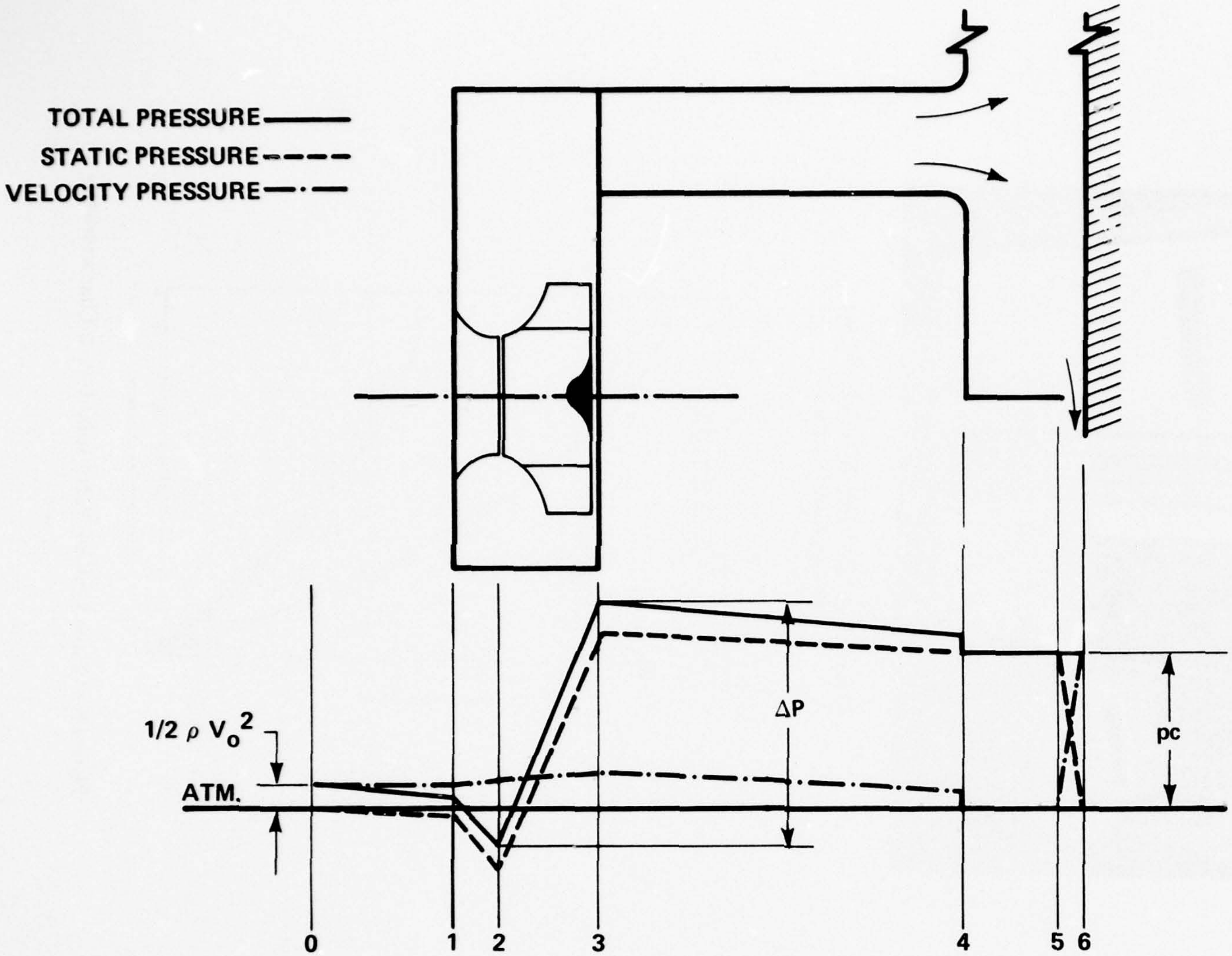

Figure 9 – SES Lift System Schematic

Inlet losses include the portion of the forward velocity head that is lost, as well as the fan inlet suction head, and are expressed as

$$H_{L_{0,2}} = (1-R_f)\frac{V_0^2}{2g} + k_2\frac{V_2^2}{2g} \tag{3}$$

The discharge loss includes that portion of the dynamic head at the fan discharge that is not recovered, expressed as

$$H_{L_{3,4}} = k_3\frac{V_3^2}{2g} \tag{4}$$

Finally, the total head rise of the fan is expressed in terms of the dimensionless head coefficient for the fan and the tip speed U_T of the impeller

$$\Delta H_{T_{2,3}} = \psi_{T_{2,3}}\frac{U_T^2}{g} \tag{5}$$

Substituting Equations (3) through (5) in Equation (2) yields

$$H_{s_4} - H_{s_0} = R_f\frac{V_0^2}{2g} + \psi_{T_{2,3}}\frac{U_T^2}{g} - k_2\frac{V_2^2}{2g} - \frac{k_2V_3^2}{g}, \tag{6}$$

Dividing each term in Equation (6) by U_T^2/g yields the following dimensionless system equation

$$\psi_{S_{4,0}} = R_f \frac{\phi_0^2}{2} + \psi_{T_{2,3}} - K_2 \frac{\phi_2^2}{2} - K_3 \frac{\phi_3^2}{2} \qquad (7)$$

where ϕ_n is the velocity of the airflow at station "n" divided by the tip velocity of the impeller.

The model fans are tested at tip speeds equal to the full-scale fan tip speed, which produces the same pressure rise for the model and full-scale fan. The range of flow perturbation frequencies is selected to envelope the range of frequencies normally encountered by a ship in a seaway. It is possible, with the flow control valve, to simulate the random flow perturbations which occur on a ship in a seaway by using an electrical signal composed of multiple discrete sinusoidal signals, or else by using a random signal generator coupled with a properly tuned filter. Mach number effects for the full-scale fan are low and are lower still for the model fan, and are therefore considered usually not to affect the performance of the fan. Similarly, the Reynolds number for the model fan is smaller than the corresponding full-scale value. The Reynolds number does affect the flow rate and efficiency for values less than $Re = 10^7$.[3]

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[3] *Balje, O.E., "A Study of Design Criteria and Matching of Turbomachines,"* Part B, Compressor and Pump Performance and Matching of Turbo Components, ASME Journal of Engineering for Power, pp. 103-114, (Jan 1962).

Figures 10 and 11 show the effect of Reynolds number on the flow rate and efficiency, respectively. Figure 10 is used for correction of flow rate in any dimensional, normalized, or dimensionless expression.

FAN EVALUATION PROCEDURE

Upon arrival of a fan model at the DTNSRDC fan evaluation rig, the fan will be mated to the electric motor. Normally, because the motor is required to operate at high speeds, a speed-reducing pulley and timing belt are used betwen the motor and the fan. In addition, instrumentation rings for flow measurement are attached to the fan. The instrumentation rings contain probes for measuring the static and total pressure at the measurement station. Transducers for measuring the raw data are calibrated and installed, and the data acquisition system software is checked out and debuggged. Experience has shown that this is a relatively time-consuming series of tasks, which usually require about two to three weeks to complete, accounting for contingencies. It is mandatory that the fan be dynamically balanced to the rotational speeds anticipated during the test.

For steady state testing, an automated procedure has been developed wherein the flow control valve is continuously swept at a very slow rate from open to closed back to open position. The data acquisition system samples the data at predetermined intervals throughout the sweep, and following the run, prints out a point-by-point summary of the reduced data parameters. Normally, at least two runs are made at different rpm's to

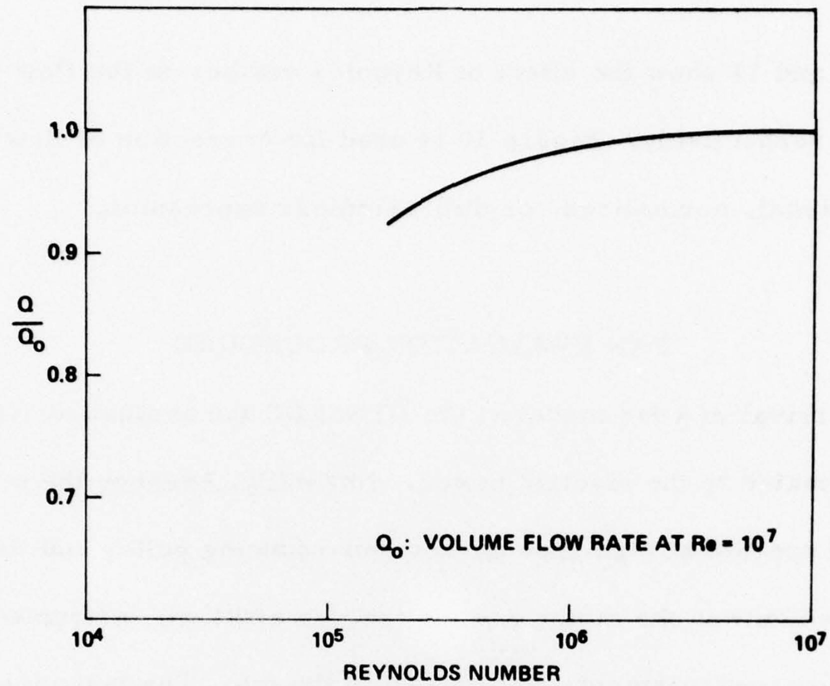

Figure 10 – Reynolds Number Effect on Flow

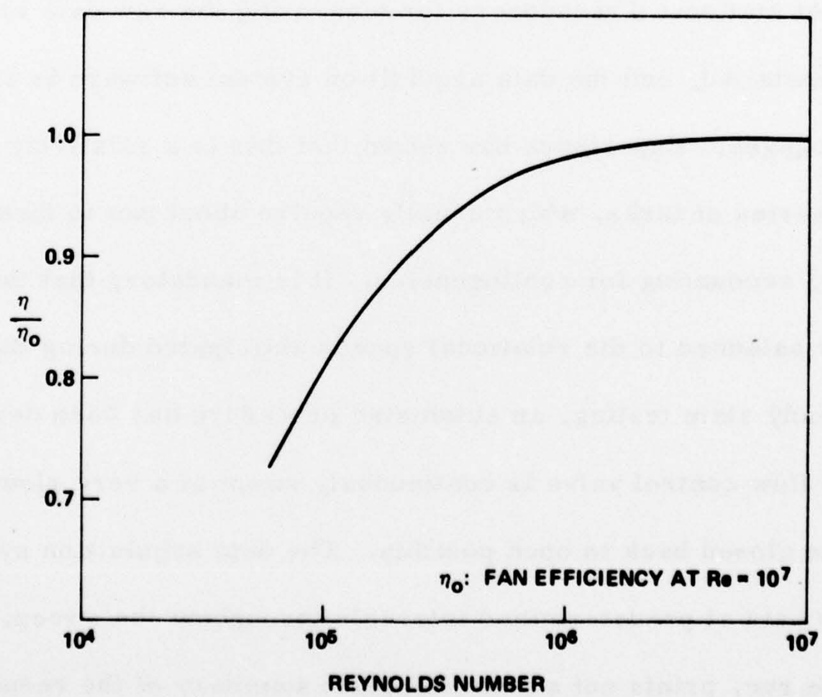

Figure 11 – Reynolds Number Effect on Efficiency

verify the fan performance. That is, a check is made to insure that the pressure, flow, and horsepower values do indeed collapse to consistent normalized values. For sinusoidal dynamic tests, the flow control valves are oscillated at a fixed amplitude about a given mean point at various frequencies. A sinusoidal signal generator provides the electrical signal to drive the valve, and various flow perturbation frequencies are established simply by adjusting the signal generator output frequency. Depending on the particular fan being tested, dynamic effects can appear at sinusoidal frequencies as low as 0.1 Hz. In general, the maximum frequency will be on the order of 8.0 Hz. Once the sinusoidal oscillation is established, the data acquisition system is started. It operates at a constant sampling rate for several seconds in order that a minimum of three complete cycles of the oscillation can be recorded. For irregular dynamic tests, an irregular electrical signal composed of a number of sinusoids at discrete frequencies and amplitudes is generated and used to drive the flow control valve around a given mean position. Usually for these tests, selected parameters are recorded on FM tape for subsequent (off-line) processing. This includes spectrum analyses, as well as statistical analysis to determine the overall mean value and standard deviation of the particular parameter, as well as the mean values of all the maxima or minima and the mean values of the 1/3 highest and 1/10 highest maxima or minima.

CONCLUSIONS

A lift fan evaluation rig has been designed, fabricated and integrated with the 8- by 10-foot subsonic wind tunnel facilities at DTNSRDC. Centrifugal and axial flow lift fans, up to about 2 feet in diameter, can be evaluated in the rig -- the primary constraint to size being the available power to drive the fan. Electric motors are available up to 150 hp, or to approximately 225 hp if two motors are used in parallel to drive a single fan. Instrumentation, consisting of pressure, temperature, torque, speed, and position transducers, is available to measure the raw data parameters. Data are recorded using a Raytheon 704 digital data acquisition system consisting of an analog to digital converter, a processing unit, and various peripheral devices. The Raytheon computer produces tabulated output data. Automatically plotted data are produced off-line using the Center's CDC computers.

ACKNOWLEDGEMENT

The Lift Fan Evaluation Program at DTNSRDC was conducted under the sponsorship of Mr. Sydney Davis of the Surface Effect Ship Project. The original design effort was undertaken by personnel of the Design Engineering Branch; Mr. Richard Dye was the project engineer for the design of the rig under the supervision of Mr. Ernie Screen. Subsequently, this project was transferred to the Surface Effect Ship Division under the cognizance of the author. The Aerojet Liquid Rocket Company under the supervision of

Mr. Werner Luscher, was tasked with the development of techniques for fan testing. The contributions of all these persons are acknowledged with gratitude.

APPENDIX

DERIVATION OF DATA REDUCTION EQUATIONS

The assumptions utilized for reducing the fan data are:

1) The water vapor - air mixture behaves as a perfect gas.

2) The duct loss coefficients are constant and independent of the Reynolds number and the Mach number.

3) Compressibility between the fan inlet and exit is included.

4) The flow process is irreversible and adiabatic.

The basic equations include the isentropic process relationship, the equation of state and the energy equation for a perfect gas. The isentropic relationships are

$$\frac{P_s}{P_T} = \left(\frac{\rho_s}{\rho_T}\right)^{\gamma} \qquad \text{(A-1)}$$

and

$$\frac{T_s}{T_T} = \left(\frac{P_s}{P_T}\right)^{(\gamma-1)/\gamma} \qquad \text{(A-2)}$$

These relationships hold at any point or station in the system. The equation of state, again valid at any point in the system, is

$$\frac{P_s}{\rho_s T_s g} = \frac{P_T}{\rho_T T_T g} = R \qquad \text{(A-3)}$$

and

$$h = C_p T \qquad \text{(A-4)}$$

And, finally, the energy equation for a perfect gas is

$$h_{T_n} = h_{s_n} + \frac{V_n^2}{2gJ} = C_p T_{s_n} + \frac{V_n^2}{2gJ} = C_p T_{T_n} \qquad \text{(A-5)}$$

$$T_{T_n} = T_{s_n} + \frac{V_n^2}{2gC_p J} \qquad \text{(A-6)}$$

$$\Delta h_n = C_p \Delta T_n = C_p T_n \left(\frac{P_{n+1}}{P_n}\right)^{(\gamma-1)/\gamma} \quad \text{(A-7)}$$

The following output parameters are calculated from the measured data at the respective stations.

Station 1 - Impeller Shaft

The shaft power is calculated from the torque and rpm using

$$SHP = \frac{L \cdot N}{5252} \quad \text{(A-8)}$$

Station 2 - Wind Tunnel Test Section

Using the wind tunnel psychrometer, the air temperature and humidity (β) is measured. The humidity is the number of pounds of water vapor per pound of dry air.* With this value of β, the gas properties are calculated using the following equations:

$$R = \frac{53.3 + 85.7\beta}{1 + \beta} \quad \text{(A-9)}$$

$$C_p = \frac{.24 + .45\,\beta}{1 + \beta} \quad \text{(A-10)}$$

$$\gamma = \frac{1}{1 - \frac{R}{C_p \cdot J}} \quad \text{(A-11)}$$

The inlet state points are determined as follows:

Static Pressure P_{s_2} : By measurement

Dynamic (Ram) Pressure ΔP_{D_2} : By measurement

Total Pressure $P_{T_2} = P_{s_2} + \Delta P_{D_2}$

Total Temperature T_{T_2} = Dry bulb temperature

*Zimmerman, O. T. and I. Lavine, "Psychrometric Tables and Charts," 2nd Edition. Industrial Research Service, Dover, New Hampshire, (1964).

$$\text{Static Temperature } T_{s_2} = T_{T_2} \left(\frac{P_{s_2}}{P_{T_2}} \right)^{(\gamma-1)/\gamma} \qquad \text{(A-12)}$$

$$\text{Static Density } \rho_{s_2} = \frac{P_{s_2}}{R \cdot T_{s_2} \cdot g} \qquad \text{(A-13)}$$

$$\text{Ram Velocity } V_2 = \frac{2g\gamma RT_{T_2}}{\gamma-1} \left[1 - \left(\frac{P_{s_2}}{P_{T_2}} \right)^{(\gamma-1)/\gamma} \right]^{\frac{1}{2}} \qquad \text{(A-14)}$$

<u>Station 3 - Fan Inlet</u> (Station 4 - Fan Discharge)

Stat: - Pressure $P_{s_{3(4)}}$: Average of at least 4 measurements

Dynamic Pressure $\Delta P_{D_{3(4)}}$: Average of at least 9 measurements

Total Temperature $T_{T_{3(4)}}$: Average of at least 3 measurements

$$\text{Static Temperature } T_{s_{3(4)}} = T_{T_{3(4)}} \left(\frac{P_{s_{3(4)}}}{P_{T_{3(4)}}} \right)^{(\gamma-1)/\gamma} \qquad \text{(A-15)}$$

$$\text{Density } \rho_{s_{3(4)}} = \frac{P_{s_{3(4)}}}{R \cdot T_{s_{3(4)}} \cdot g} \qquad \text{(A-16)}$$

$$\text{Velocity } V_{3(4)} = \frac{2g\gamma RT_{T_{3(4)}}}{(\gamma-1)} \left[1 - \left(\frac{P_{s_{3(4)}}}{P_{T_{3(4)}}} \right)^{(\gamma-1)/\gamma} \right]^{\frac{1}{2}} \qquad \text{(A-17)}$$

$$\text{Volume Flow Rate } Q_{3(4)} = V_{3(4)} A_{3(4)} \qquad \text{(A-18)}$$

$$\text{Weight Flow Rate } W_{3(4)} = \rho_{s_{3(4)}} Q_{3(4)}\, g \qquad \text{(A-19)}$$

Flow Coefficients

$$U_{TIP} = \frac{N\ d_t}{229} \qquad \text{(A-20)}$$

$$\phi_2 = \frac{V_2}{U_{TIP}} \qquad \text{(A-21)}$$

$$\phi_3 = \frac{V_3}{U_{TIP}} \qquad \text{(A-22)}$$

$$\phi_4 = \frac{V_4}{U_{TIP}} \qquad \text{(A-23)}$$

$$\phi = \frac{Q_3}{A_R\ U_{TIP}} \qquad \text{(A-24)}$$

System Head Rise

$$\Delta H_{s_{(2-4)}} = C_p J\ T_{s_2} \left[\left(\frac{P_{s_4}}{P_{s_2}} \right)^{(\gamma-1)/\gamma} - 1 \right] \qquad \text{(A-25)}$$

Fan Head Rise

Total/Total

$$\Delta H_{T_{(3-4)}} = C_p J\ T_{T_3} \left[\left(\frac{P_{T_4}}{P_{T_3}} \right)^{(\gamma-1)/\gamma} - 1 \right] \qquad \text{(A-26)}$$

-Total/Static

$$\Delta H_{S_{(3-4)}} = C_p J\ T_{s_3} \left[\left(\frac{P_{s_4}}{P_{T_3}} \right)^{(\gamma-1)/\gamma} - 1 \right] \qquad \text{(A-27)}$$

Torque Coefficient

$$\tau = \frac{\phi\ \psi_{T_{3-4}}}{\eta_T} \qquad \text{(A-28)}$$

Head Coefficients

$$\psi_{s_{2-4}} = \frac{\Delta H_{s_{2-4}} g}{U_{TIP}^2} \tag{A-29}$$

$$\psi_{T_{3-4}} = \frac{\Delta H_{T_{3-4}} g}{U_{TIP}^2} \tag{A-30}$$

$$\psi_{s_{3-4}} = \frac{\Delta H_{s_{3-4}} g}{U_{TIP}^2} \tag{A-31}$$

Head Losses and Loss Coefficients

$$H_{L_{2-3}} = C_p \; J \; T_{T_2} \left[\left(\frac{P_{T_3}}{P_{T_2}} \right)^{(\gamma-1)/\gamma} - 1 \right] \tag{A-32}$$

$$k_3 = \frac{2H_{L_{2-3}} g}{V_3^2} \qquad \text{evaluated at } V_2 = 0. \tag{A-33}$$

$$H_{L_{4-8}} = C_p \; J \; T_{T_4} \left[\left(\frac{P_{T_4}}{P_{s_4}} \right)^{(\gamma-1)/\gamma} - 1 \right] \tag{A-34}$$

$$k_4 = \frac{2H_{L_{4-8}} g}{V_4^2} \tag{A-35}$$

Ram Recovery Factor

$$R_f = 1 + k_3 \left(\frac{V_3}{V_2} \right)^2 - \frac{2H_{L_{2-3}} g}{V_2^2} \tag{A-36}$$

Fan Efficiency

- Total/Total

$$\eta_T = \frac{W_3 \, \Delta H_{T_{3-4}}}{550 \text{ SHP}} \tag{A-37}$$

$$\text{or } \eta_T = \frac{T_{T_2}\left[\left(\dfrac{P_{T_4}}{P_{T_3}}\right)^{(\gamma-1)/\gamma} - 1\right]}{T_{T_4} - T_{T_2}} \tag{A-38}$$

- Total/Static

$$\eta_s = \frac{W_3 \, \Delta H_{s_{3-4}}}{550 \text{ SHP}} \tag{A-39}$$

Standard Inlet Conditions

The dimensional fan characteristics are corrected for standard inlet conditions, viz.

$$T_{T_3} = 59 \text{ F}$$

$$\text{Humidity} = 0.00535 \ \frac{\text{lb } H_2O}{\text{lb air}} \ \text{(50-Percent Relative Humidity)}$$

$$P_{T_3} = 14.696 \text{ psia} = 2116 \text{ psfa}$$

$$\gamma = 1.3987$$

$C_p = 0.2411$ Btu/lb F

$R = 53.47$ ft lb/lb

The pressure rise from the fan inlet to the discharge (total/total) is:

$$\Delta P_{T_{3,4}} = P_{T_3}\left(\frac{P_{T_4}}{P_{T_3}} - 1\right) \tag{A-40}$$

From the dimensionless ϕ, ψ characteristics, the pressure ratio is

$$\frac{P_{T_4}}{P_{T_3}} = \left(1 + \frac{\psi_T\, U_{TIP}^{2}}{g\, C_p\, J\, T_{T_3}}\right)^{\gamma/(\gamma-1)} \tag{A-41}$$

and the volume flow rate is

$$Q_3 = \frac{\pi\, d\, b}{144}\, U_{TIP}\, \phi. \tag{A-42}$$

Choosing a design point, for instance, the ϕ, ψ values corresponding to maximum efficiency, and a design tip speed, one can then express the dimensional pressure-flow characteristics from these as

$$\Delta P_{T_{3-4}} = P_{T_3}\left\{\left[1 + \left(\frac{\psi_{T_{3-4}D}\, U_D^{2}}{g\, C_p\, J\, T_{T_3}}\right)\left(\frac{\psi_{T_{3-4}}}{\psi_{T_{3-4}D}}\right)\left(\frac{U}{U_D}\right)^{2}\right]^{\frac{\gamma}{\gamma-1}} - 1\right\} \tag{A-43}$$

$$\text{and}\quad Q = \left(\frac{\pi\, d_4\, b_4}{144} * U_{TIP_D} * \phi_D\right)\left(\frac{U_{TIP}}{U_{TIP_D}}\right)\left(\frac{\phi}{\phi_D}\right) \tag{A-44}$$

The above procedure is repeated to calculate the fan total-to-static head rise by replacing P_{T_4} by P_{s_4}, and ψ_T by ψ_s and T by s in Equations (A-40) through (A-44).